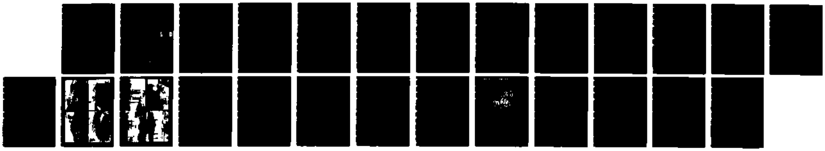
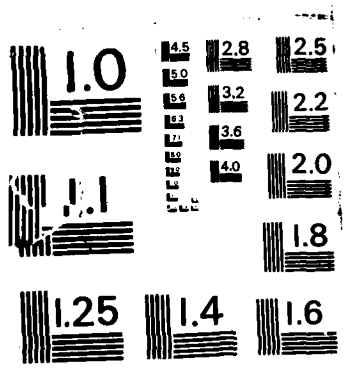
AD-A167 401 INSPECTION OF THE PROPULSION SYSTEM OF THE BARGE SEACON 1/1
(U) GIANNOTTI AND ASSOCIATES INC ANNAPOLIS MD
P R VAN MATER 18 OCT 79 GA-78-026-002
UNCLASSIFIED CHES/NAVFAC-FPO-7916 N62477-78-C-0346 F/G 13/10 NL





MICROCOPY RESOLUTION TEST CHART
NATIONAL BUREAU OF STANDARDS-1963-A

OTTI & ASSOCIATES, INC.

NAVAL ARCHITECTS
OCEAN ENGINEERS
MARINE ENGINEERS

Chern
10-23-79

1

INSPECTION OF THE PROPULSION SYSTEM OF THE BARGE "SEACON"

G&A REPORT NO. 78-026-002

prepared by

PAUL R. VAN MATER, JR.

for

NAVAL FACILITIES ENGINEERING COMMAND
CHESAPEAKE DIVISION (FPO-1)

under

CONTRACT N62477-78-C-0346

18 OCTOBER 1979

703 GIDDINGS AVENUE, SUITE U-3 • ANNAPOLIS, MARYLAND 21401 • (301) 268-0030 • D.C. 261-1031

Unclassified
SECURITY CLASSIFICATION OF THIS PAGE

REPORT DOCUMENTATION PAGE

1a. REPORT SECURITY CLASSIFICATION
Unclassified

1b. RESTRICTIVE MARKINGS

2a. SECURITY CLASSIFICATION AUTHORITY

3. DISTRIBUTION AVAILABILITY OF REP.
Approved for public release; distribution is unlimited

2b. DECLASSIFICATION/DOWNGRADING SCHEDULE

4. PERFORMING ORGANIZATION REPORT NUMBER
G&A Report No. 78-026-002

5. MONITORING ORGANIZATION REPORT #
FPO 7916

6a. NAME OF PERFORM. ORG.
Giannotti & Associates, Inc.

6b. OFFICE SYM

7a. NAME OF MONITORING ORGANIZATION
Ocean Engineering
& Construction
Project Office
CHESNAVFACENGCOM

6c. ADDRESS (City, State, and Zip Code)
793 Giddings Ave., Suite U-3
Annapolis, MD 21401

7b. ADDRESS (City, State, and Zip)
BLDG. 212, Washington Navy Yard
Washington, D.C. 20374-2121

8a. NAME OF FUNDING ORG.

8b. OFFICE SYM

9. PROCUREMENT INSTRUMENT INDENT #
N62477-78-C-0346

8c. ADDRESS (City, State & Zip)

10. SOURCE OF FUNDING NUMBERS

PROGRAM ELEMENT #	PROJECT #	TASK #	WORK UNIT ACCESS #

11. TITLE (Including Security Classification)
Inspection of the Propulsion System of the Barge "Seacon"

12. PERSONAL AUTHOR(S)

13a. TYPE OF REPORT

13b. TIME COVERED
FROM TO

14. DATE OF REP. (YYMMDD)
79-10-18

15. PAGES
19

16. SUPPLEMENTARY NOTATION

17. COSATI CODES

FIELD	GROUP	SUB-GROUP

18. SUBJECT TERMS (Continue on reverse if nec.)
Seacon, Barges, Propulsion systems

19. ABSTRACT (Continue on reverse if necessary & identify by block number)
Results of an analysis of the powering losses in the propulsive system of the barge SEACON are reported together with the results of an inspection trip to examine the SEACON propulsive components while the barge was in drydock. Findings are that engine condition contributed to poor speed (Con't)

20. DISTRIBUTION/AVAILABILITY OF ABSTRACT
SAME AS RPT.

21. ABSTRACT SECURITY CLASSIFICATION

22a. NAME OF RESPONSIBLE INDIVIDUAL
Jacqueline B. Riley

22b. TELEPHONE
202-433-3881

22c. OFFICE SYMBOL

DD FORM 1473, 84MAR

SECURITY CLASSIFICATION OF THIS PAGE

BLOCK 19 (Con't)

performance before engine replacement , and that the inherently low efficiency of the Voith-Schneider propulsors contribute significantly to the losses. A power budget is presented which identifies each loss component.

SUMMARY

Results of an analysis of the powering losses in the propulsive system of the barge SEACON are reported together with the results of an inspection trip to examine the SEACON propulsive components while the barge was in drydock. Findings are that engine condition contributed to poor speed performance before engine replacement, and that the inherently low efficiency of the Voith-Schneider propulsors contribute significantly to the losses. A power budget is presented which identifies each loss component.

TABLE OF CONTENTS

1.0 BACKGROUND

In Reference 1* results of an experimental study of a directional stability problem associated with the barge SEACON were reported. Also conducted in connection with this study were the results of resistance tests of a model of the barge. Using hearsay information on the speed performance of the barge and a number of assumptions, the resistance penalties associated with various fixes to the directional stability problem were assessed. In the course of this assessment, it was noted that the Propulsive Coefficient inferred from these results, P.C. = .134 at 7.0 knots, was very poor. During the recent drydocking of the barge at Bellinger Shipyard, Jacksonville, Florida, an opportunity presented itself to inspect the hull and Voith-Schneider propellers while out of water to seek an explanation of the poor behavior. Accordingly, Dr. Paul R. Van Mater, Jr., visited the SEACON in the yard on 7 June 1979 while the ship was drydocked. This report documents the results of that visit. Further information on the propulsive behavior was provided as a result of the measured mile speed trials which were conducted following the shipyard period. With new aft engines installed and with reconditioned V-S propulsors the barge achieved 7.9 knots on trials.

2.0 POSSIBLE EXPLANATIONS OF PROPULSIVE BEHAVIOR

Prior to shipyard modification, the propulsion plant consisted of one Detroit Diesel 12V-71 driving the forward V-S unit and two 6V-71's

*(1) "Directional Stability Model Tests of the Barge 'SEACON'," Van Mater, P. R., Jr., and Stambaugh, K. A., Giannotti & Associates, Inc., Report No. 78-026-001 for Naval Facilities Engineering Command, 31 January 1979.

driving each of the two aft V-S units for a total of 1,020 installed BHP. At a 7.75' x 9.25' draft the model test results indicated that at 7.0 knots, an observed speed condition with clean bottom, the barge requires (according to model tests) 130 Effective Horsepower (EHP) to propel it. Under recent operating experience only 5.2 knots have been experienced in day-to-day operations. The following are possible explanations or poor propulsive behavior inferred from these observations.

a) Inaccuracy in model test

b) Badly fouled hull bottom

c) Damaged or fouled V-S blades

d) Improper design or selection of V-S blading

e) High frictional losses in V-S gearing and linkages

f) High losses in reduction gears

g) Poor engine condition

h) Poor information on performance

Each of the above will be addressed in Section 3.0. Inferences on the most probable causes will be made in Section 4.0.

3.0 DISCUSSION AND INSPECTION TRIP RESULTS

3.1 MODEL TESTS

The resistance model tests reported in Table III of Reference (1) and reproduced here as Table I, were conducted for the purpose of evaluating the resistance penalty associated with various directional stability fixes, not for the purpose of establishing the

TABLE I

MODEL AND PROTOTYPE RESISTANCE AND EHP's

- - -Model- - -		- - - - - - -Prototype- - - - - - -			
v_m ft/sec	R_{t_m} lbs	V_s knots	$\bar{R}_{t_s}$ smooth lbs	EPHs clean bot.	EPHs fouled bot.
		7.75' x 9.25' DRAFT			
		0° skeg angle, no flaps			
1.19	.118	3.99	2320	28	50
1.80	.238	6.03	4697	87	163
2.12	.320	7.10	6403	140	266
2.95	.637	9.88	13800	418	770
3.66	1.100	12.26	25999	978	1661
		0° skeg angle, flaps			
1.50	.161	5.01	2996	46	89
2.01	.279	6.73	5400	112	219
2.41	.422	8.07	8778	218	406
3.53	1.040	11.83	24634	894	1506
4.38	2.048	14.67	53928	2429	3616
		15° skeg angle, no flaps			
1.59	.220	5.33	4713	77	129
2.37	.400	7.94	8190	200	379
3.03	.880	10.15	21603	673	1055
3.69	1.460	12.36	37927	1439	2140
4.48	2.570	15.01	70866	3264	4537
		15° skeg angle, flaps			
1.49	.220	4.99	4989	76	119
2.00	.385	6.70	8990	185	291
3.00	.920	10.05	23077	712	1082
4.00	1.900	13.40	51070	2100	2999
		20° skeg angle, flaps			
1.97	.499	6.60	12914	262	363
3.08	1.120	10.32	29433	932	1334
3.75	1.720	12.56	46344	1787	2523
		25° skeg angle, flaps			
1.63	.460	5.46	12652	212	268
2.59	.900	8.68	24120	642	878
3.07	1.240	10.29	33504	1057	1455
3.95	2.210	13.23	61739	2507	3372
4.46	3.060	14.94	87423	4009	5264
		10' x 12' DRAFT			
		0° skeg angle, flaps			
2.02	.467	6.77	11323	235	354
3.07	.940	10.29	22725	717	1150
3.50	1.305	11.73	32738	1180	1828
3.99	1.900	13.37	50012	2052	3021

* Assuming Coefficient of Friction is constant and equals 5.00×10^{-3}

basic resistance signature of the barge. Thus, only four or five points were taken at each condition whereas in a standard resistance test at least a dozen to fifteen points would have been taken. Following standard procedures the resistance tests were conducted with the V-S model units removed and replaced with fairing pieces. The reason for this is that the drag of the operating propulsor unit is reflected in the efficiency of the propulsor. For the case of the barge under tow with the V-S units stopped the measured resistance of the barge will not include the drag of the V-S unit and will be low by an estimated one to two percent.

Considering the abreviated nature of the model tests the question is: If the barge model were to be re-subjected to a full resistance test series, how much difference would there be in the results? To make a somewhat subjective evaluation of the quality of the model test information a comparison is made in Figure 1 of the expanded SEACON model test data with:

a) Results of a CUSS I model test at Stevens Institute reported in Reference (2). CUSS I is a floating drilling barge converted from a Navy YFNB.

b) Estimates of the resistance of a barge of similar displacement and block coefficient using the method of Moss and Townsend in Reference (3). This method uses Residuary Resistance Coefficient based composite plots of model test data. Computational details are shown in Table II.

(2) "Drilling from a Floating Vessel and the CUSS I," Bauer, R. F., Crooke, R. C., Stratton, H., Northern/Southern California Section, SNAME, October 10 & 11, 1958.

(3) "Design Considerations and the Resistance of Large Towed Seagoing Barges," Moss, J. L. and Townsend, C., III, Technical and Research Bulletin No. 1-29, SNAME, May 1969.

Table II

SEACON RESISTANCE ESTIMATE

from "Design Considerations and the Resistance of Large Seagoing Towing Barges"

James L. Moss and Corning Townsend, III
T&R Bulletin No. 1-29, SNAME

SEACON Characteristics at Normal Draft

L	= 260 Ft.
B	= 48 Ft.
T_f	= 9.25 Ft.
T_u	= 7.75 Ft.
T_m	= 8.50 Ft.
Δ	= 2,351 Ft.
Wetted Surface	= 14,778 Ft.
C_B	= .776 Ft.
ν	= $1.2817 \times 10^{-5}\ \frac{ft^2}{sec}$
ρ	= $1.9905\ \frac{lb/sec^2}{ft^4}$
$\frac{\rho}{2}S$	= 14,708

C_R Values Extrapolated from Figure 9

$\frac{V_K}{\sqrt{L}}$	$C_R \times 10^3$ Low	High	Mean
.3	1.32	1.48	1.40
.4	0.91	1.15	1.03
.5	0.85	1.07	0.96

Computation

							LOW				HIGH			
$\frac{V_K}{\sqrt{L}}$	V_K	$.915\sqrt{K}$ *	v	$R_N \times 10^{-8}$	C_f	C_A	C_R	C_T	R_T	EHP	C_R	C_T	R_T	EHP
.3	4.84	4.42	8.165	1.656	1.939	.4	1.32	3.659	3588	53	1.48	3.819	3744	56
.4	6.450	5.90	10.887	2.209	1.863	.4	0.91	3.173	5531	109	1.15	3.413	5950	118
.5	8.062	7.38	13.609	2.760	1.809	.4	0.85	3.059	8333	206	1.07	3.279	8932	221

* Effect of skegs treated as an 8.5% speed loss.

The agreement between the model test data and the other sources shown in Figure 1 is fair, but the SEACON results appear, if anything, a little low. The test drafts, displacement, and configuration details of the CUSS I model test are not well documented in Reference (2); consequently, the displacement identified in Figure 1 has some uncertainty associated with it. The Moss-Townsend prediction is presented as a band using composite barge data of uncertain applicability. At the same time the SEACON points, although few in number, do represent measured data under known conditions. Weighing all this we will increase the SEACON EHP lines by 15% to account for data paucity and possible scaling errors.

Figure 1 also shows a line labeled "Best Estimate, Trial Condition, Δ = 2650 LT." This curve was developed by interpolating between the Δ = 2351 LT line and the Δ = 3441 LT line for the estimated trial displacement, Δ = 2650 LT, then applying the 15% allowance discussed above. This gives us our best estimate of the EHP at the trial speed of 7.90 knots as 270 horsepower.

3.2 HULL BOTTOM

Visual examination of the SEACON while in drydock showed the hull bottom to be in excellent condition with only occasional spots of very minor barnacle accumulation. Bottom fouling could not have been a factor in speed loss.

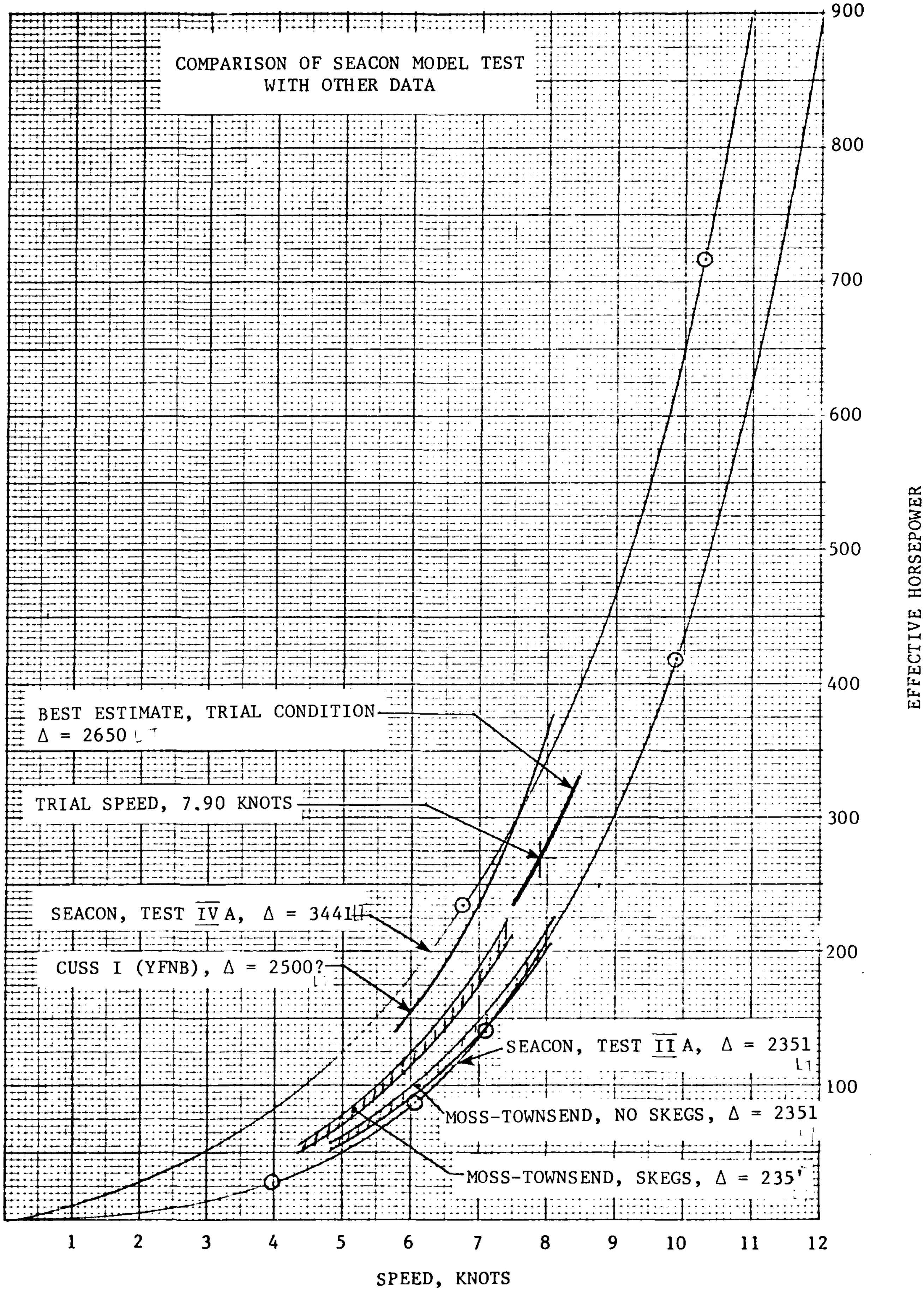

COMPARISON OF SEACON MODEL TEST
WITH OTHER DATA
BEST ESTIMATE, TRIAL CONDITION
Δ = 2650
TRIAL SPEED, 7.90 KNOTS
SEACON, TEST IV A, Δ = 3441
CUSS I (YFNB), Δ = 2500?
SEACON, TEST II A, Δ = 2351
MOSS-TOWNSEND, NO SKEGS, Δ = 2351
MOSS-TOWNSEND, SKEGS, Δ = 235
900
800
700
600
500
400
300
200
100
EFFECTIVE HORSEPOWER
1
2
3
4
5
6
7
8
9
10
11
12
SPEED, KNOTS

Figure 1.

3.3 V-S BLADES

All but five of the V-S blades had been removed at the time of the inspection. Figure 2 shows several views of the stern at the time of inspection. One of the remaining blades had slight damage, as can be seen in the upper right view of Figure 2. Blades removed from the forward V-S unit had suffered some damage, apparently from an earlier tow cable entanglement. Views of four of these blades are shown in Figure 3. The damage observed could account for some loss in efficiency, but probably less than 10 percent.

3.4 V-S PROPULSIVE EFFICIENCY

Discussions with Mr. Dieter Stumpp of the Voith Company indicated that the blades had been specifically designed for the SEACON application and were not, as had been suspected, simply converted from a previous installation. The Voith design data sheet for the application is included as Figure 4 and shows a maximum hydrodynamic design efficiency of .80 based on a blade rotation rate of 340 RPM and an input RPM of 600. The units are designed for a thrust of 3,000 kg (6,614 lbs.) each at 11 knot running speed and 3,500 kg (7,716 lbs.) each bollard thrust. Design information on vertical axis propellers is, in general, proprietary; but what information is available in the open literature suggests that this figure is optimistic. Figures 5

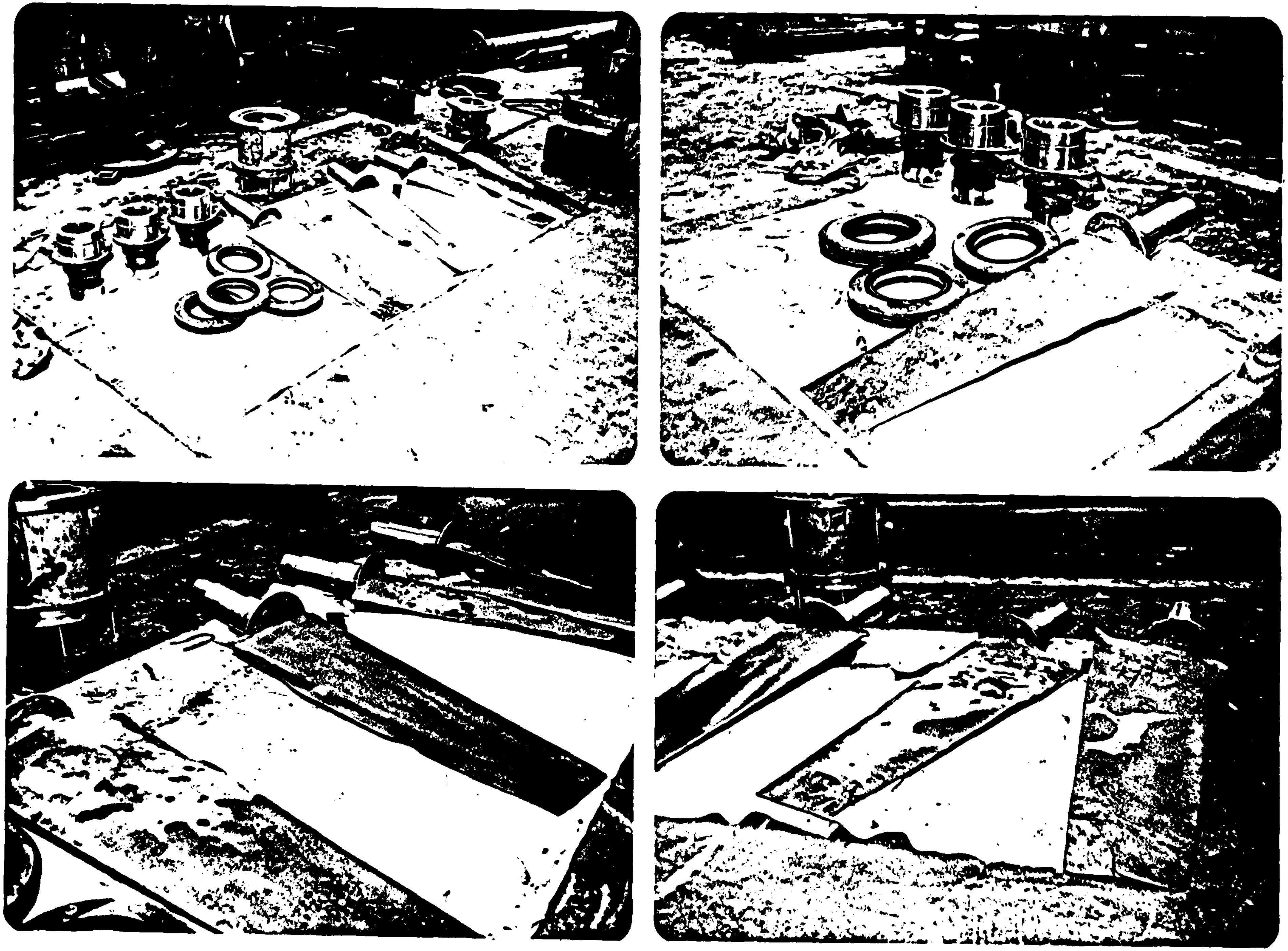


FIGURE 2. REMOVED V-S BLADING, SHOWING DAMAGE

FIGURE 3. SKEGS AND V-S UNITS IN DRYDOCK

Kennwort: VSP Landing II

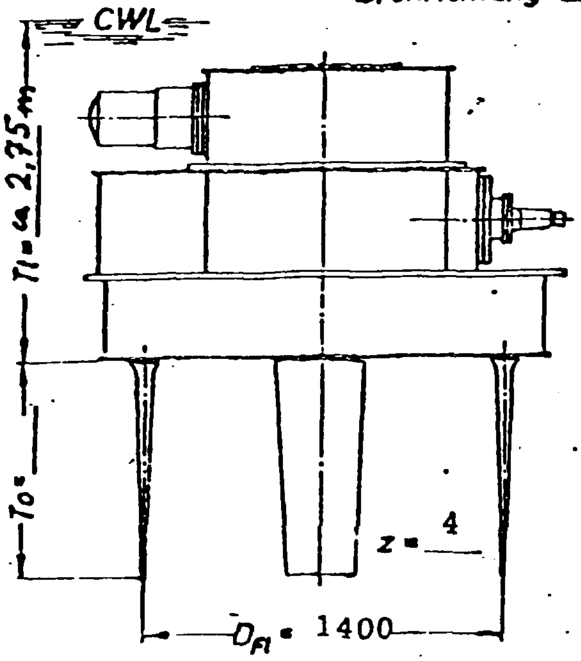
endg. vorl. **Datenzettel**

Projekt Nr.: 9-75 B.Z.Nr.: 743 838

ausgestellt am : 13.6.1975
durch : Krüger
Büro - Endtermin :
Werkstatt - Endtermin :
Versandtermin : Ende April 1976

Propeller

1 VSP Gr.: 14E/90 Prop. Nr.: 2089
Drehrichtung: rechts (wie VSP Nr. 1268)



Leistung N = 340 PS
Drehzahl Ritzel n_R = 600 min^{-1}
Drehzahl Läufer n_L = 140 min^{-1}
i = 4,27

Kinematik: SKK 1 (wie VSP Landing)
max. Steigung (Fahrt) $\lambda_{max.}$ = 0,8
errechn. Steigung (") λ_R = 0,78
max. Steig. (Ruder) =
Fahrdiagramm Nr. 3

Flügeltype: 4 M 1,5/0,25/0,625/ 1/3
Fahrt-Schub = 3 000 kg bei 11 kn(Km/h)
Stand-Schub = 3 500 kg

Steuerzeiten: voll voraus - voll zurück 6 sec.; hart - hart 5 sec.
Ölstau: ja Abnahme: ABS
Sonstiger Lieferumfang:
Bemerkungen: ~~XXXXXXXXXXXXXX~~e! Flügellagerung mit Hebelhülsen, konstruktive Ausführung wie VSP Landing
Turnvorrichtung:

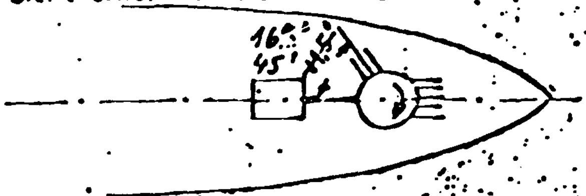
Antrieb

Motortype: Drehzahl n = 600 min^{-1}
Kraftübertragung:

Anordnung und Strahlrichtung

siehe untenstehende Skizze

Schiffstype:
Propellerneigung längs:
quer:



Fremd angetriebene Reserve-Ölpumpe, Öldruckanzeigegerät mit Anschluss für Warnanlage vorgesehen!

endgültige Lage der Ritzelwelle wird noch überprüft

Figure 4.

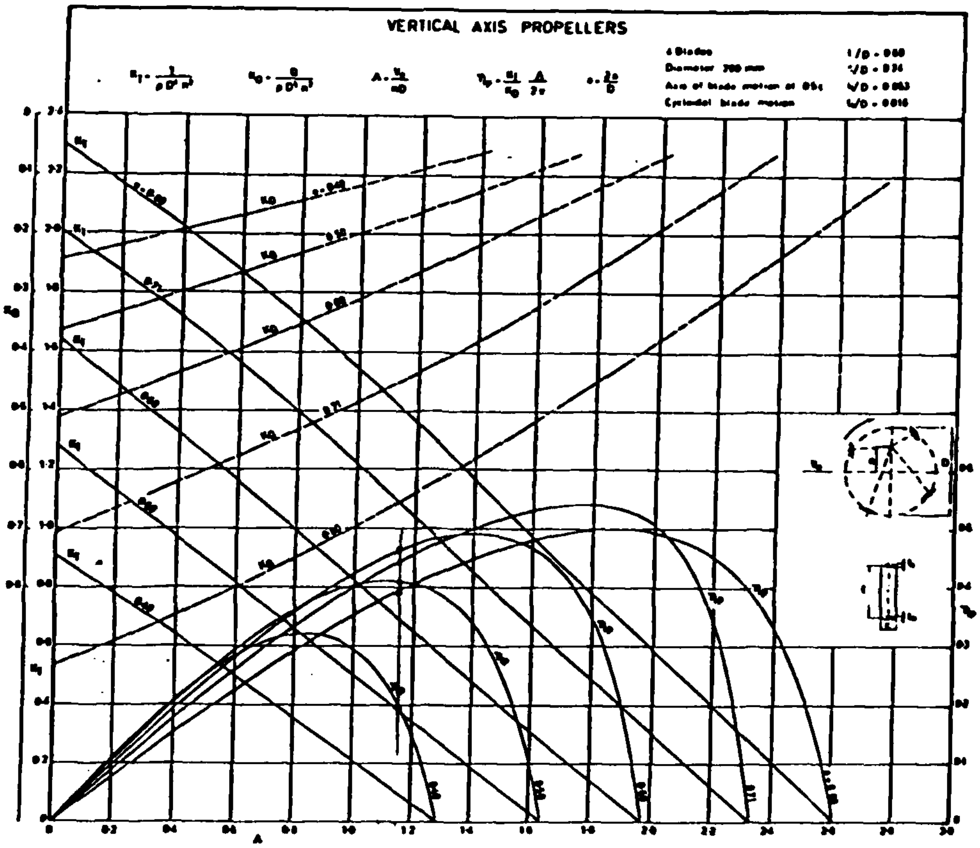


Figure 5. Open water test results for a four-bladed series with cycloidal blade motion (c/l = 0.40).

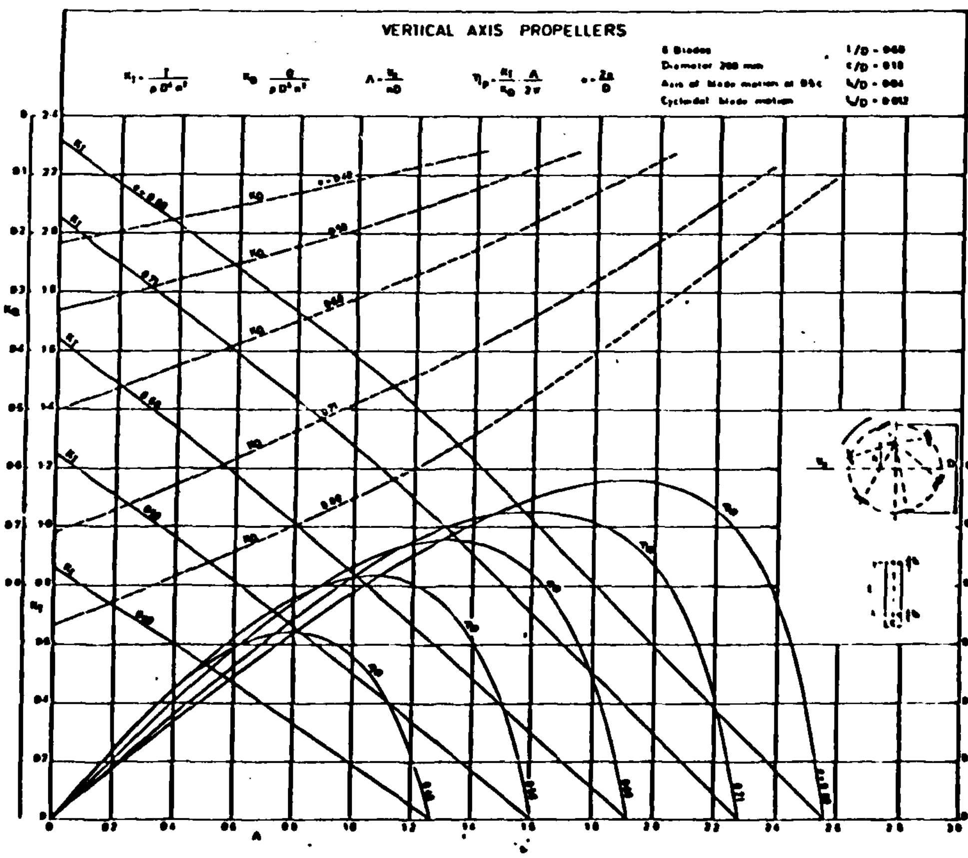


Figure 6. Open water test results for a six-bladed series with cycloidal blade motion (c/l = 0.30).

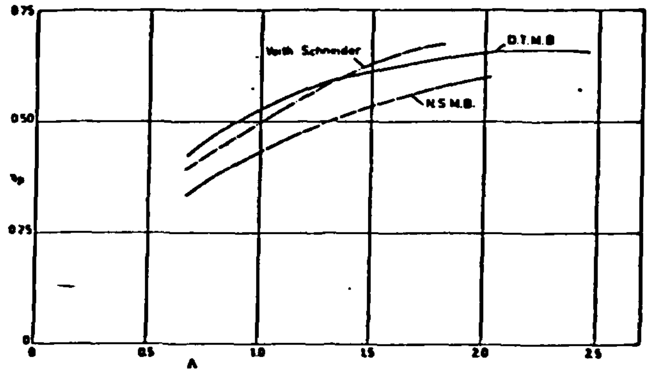


Figure 7. Optimum propeller efficiency of some vertical axis propellers.

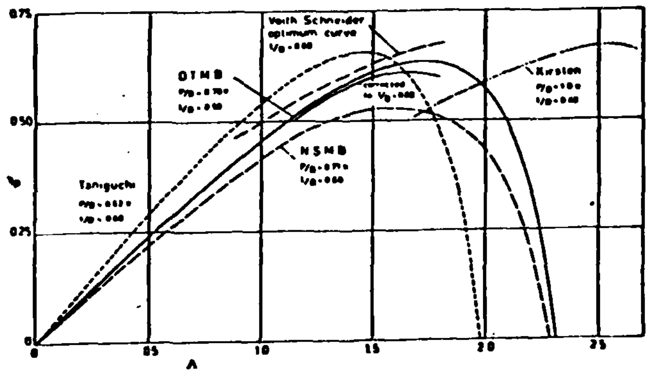


Figure 8. Propeller efficiency of compared vertical axis propellers.

and 6 taken from Reference 4* and Figures 7 and 8 taken from Reference 5* suggest upper bounds more in the range of .70 with lower average values. Consider, for example, the curves of Figure 5. These are not the curves for the SEACON installations, but it may be assumed that they are fairly similar. Three sets of curves are shown: Thrust Coefficient, K_T; Torque Coefficient, K_u; and Open Water Efficiency, η_p. The parameter in each case is the eccentricity, e, and fine values are shown. The abscissa is the Advance Coefficient, $\Lambda = \frac{Ve}{nD}$, where V_e is the Entrance Velocity, n is the revolutions per second, and D is the Blade Circle Diameter. For the SEACON trial speed of 7.9 knots assuming a Wake Fraction of .92,

$$\Lambda = \frac{.92 \times 7.9 \times 1.688}{2.33 \times 4.59} = 1.15$$

Examining the curves at $\Lambda = 1.15$ we find that the Open Water Efficiency varies from about .20 to about .46. Maximum thrust occurs at maximum eccentricity, e = .80, and at this loading $\eta_p = .39$. Again, these figures are not represented as being the correct values for the SEACON units, still they are suggestive of the range that can be expected.

*(4) "Results of Systematic Tests with Vertical Axis Propellers," van Manen, J.D., International Shipbuilding Progress, Vol. 13, No. 148, December 1966.

*(5) "A Comparison of Some Published Results of Tests on Vertical Axis Propellers," Ruys, A.W., International Shipbuilding Progress, Vol. 13, No. 148, December 1966.

3.5 V-S GEARING LOSSES

Conversations with Mr. Stumpp and published results indicate that typical mechanical losses in the V-S gearing and linkage are about 10 percent. Considering the multiple pivots, bearing surfaces and gearing, this seems to be a low value. Perhaps a value of 15 percent would be more realistic.

3.6 REDUCTION GEAR LOSSES

Losses in a reduction gear unit with two input shafts and one output shaft, as in the case of the after units, are somewhat higher than those incurred in a one in-one out gear box, as in the case of the forward unit. Losses of 10 percent and 7 percent respectively would be representative.

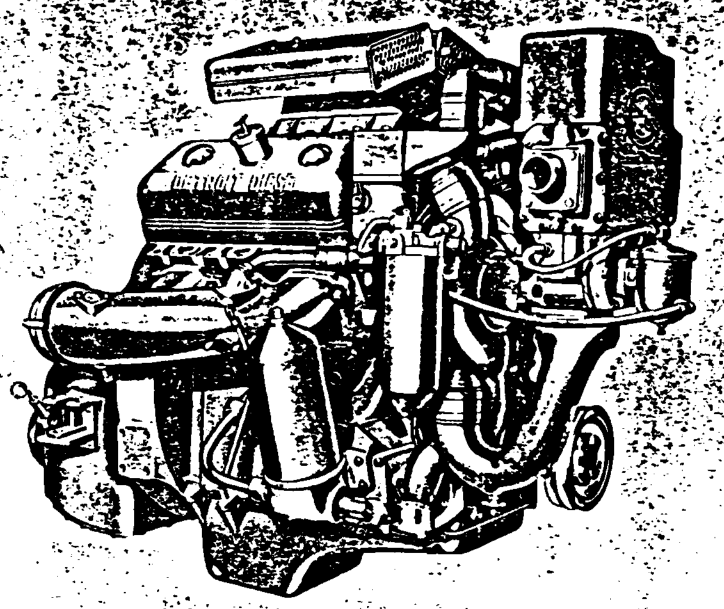
3.7 ENGINE CONDITION

No new information on this subject was obtained on the inspection trip; however, previous discussions with the Chief Engineer have indicated that the aft engines could not be operated over 1,400 RPM without overheating. Figure 8 is the specification sheet for a 6V-71 model engine. Dropping from 1,800 to 1,400 RPM would cause a loss in power from 174 BHP to 143 BHP. If, in addition, the engine compression has deteriorated due to wear, a further decline would be involved, perhaps another 15 percent.

Detroit Diesel Engines

marine models

6V-71
265 hp



Model
6V-71 7062-3000
7062-7000

Typical 6V-71
Marine Model

specifications

Basic Engine	6V-71
Model	7062-3000 (port) 7062-7000 (starboard)
Engine Type	Two Cycle
Number of Cylinders	6
Bore and Stroke	4¼ in. x 5 in.
Two Cycle Displacement (Every Downstroke a Powerstroke)	426 cu. in.
Rated Brake Horsepower 60°F and Sea Level	265 @ 2300 RPM
Rated Shaft Horsepower 85°F and 500 ft.	240 @ 2300 RPM
Continuous Shaft Horsepower	174 @ 1800 RPM
Compression Ratio	18.7 to 1
Approx. Net Weight (dry) with Standard Equipment	2570 lbs.

Rating Explanation

RATED BRAKE HORSEPOWER—Approximate basic engine power at conditions of 60°F and Sea Level.

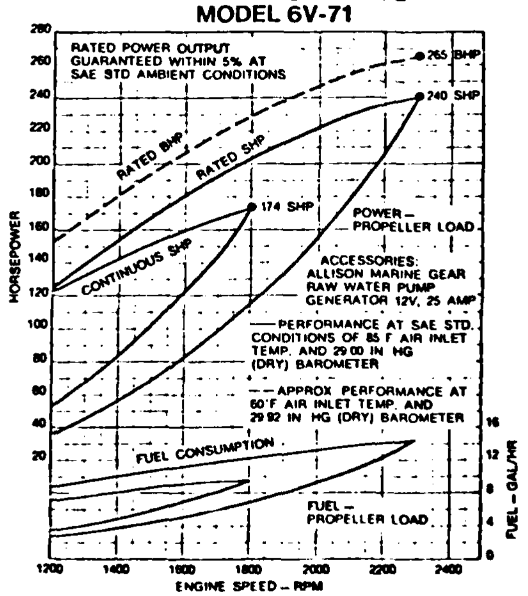
RATED SHAFT HORSEPOWER—Net power available at the marine gear output shaft; this rating is recommended for pleasure craft applications.

CONTINUOUS SHAFT HORSEPOWER—Net power available at the marine gear output shaft for continuous duty or workboat applications.

PROPELLER LOAD—Indicates horsepower absorbed by a typical propeller and the corresponding fuel consumption throughout the speed range.

Propeller load and shaft horsepowers as shown are based on ambient conditions of 85°F, Bar. (dry) 29.00 in. HG and include deduction for standard marine accessory equipment.

ENGINE PERFORMANCE*
MODEL 6V-71



*"Rated" curves based on use of N70 Injectors
"Continuous" curves based on use of N55 Injectors

Figure 8

3.8 OTHER LOSSES

Two other losses will be discussed which are present in ship propulsion systems.

a) Hull Efficiency related the thrust augment required to overcome the low pressure region created forward of the propulsor to the wake velocity recovery from entrained water carried along by the ship.

$$\eta_H = \frac{1 - t}{1 - w}$$

t = Thrust Deduction Factor. For this application, $t = .17$

w = Wake Fraction. For this application, $w = .08$

$$\eta_H = \frac{1 - .17}{1 - .08} = .90$$

b) Relative Rotative Efficiency accounts for the difference in efficiency of a propulsor as tested in a water tunnel and installed on ship. For our application,

$$\eta_{RR} = .98$$

3.9 POOR INFORMATION

Much of the information used is based on the recollections of the Captain and Chief Engineer which, although very helpful, still is not well documented. For example, how carefully were the 7 knot trials conducted? Were the aft engines operating at full RPM? What were the drafts at the time of this test? Uncertainty in these elements makes a precise analysis of the power losses before overhaul impossible.

4.0 RECAPITULATION AND POWER BUDGET

Based on the foregoing discussion, the following chain of efficiencies is proposed as a reasonable representation of conditions prevailing before and after overhaul.

Table III

	Before	After
Engine Condition, η_{ENG}	.85	1.0
Reduction Gear, η_{RG}	.91	.91
V-S Linkage, η_{LINK}	.85	.85
Blade Condition, η_{BC}	.90	1.0
Propulsor Efficiency, η_{P}	.34	.39
Hull Efficiency, η_{H}	.90	.90
Relative Rotative Efficiency, η_{RR}	.98	.98

The Propulsive Coefficient will be:

$$\text{P.C.} = \eta_{ENG} \times \eta_{RG} \times \eta_{LINK} \times \eta_{BC} \times \eta_{P} \times \eta_{H} \times \eta_{RR}$$

	Before	After
Propulsive Coefficient, P.C.	.18	.27

These results can also be represented as losses, or as a power budget as shown in Table IV.

Table IV

	7.0 knots Before Overhaul		7.9 knots After Overhaul	
	%	HP	%	HP
Rated BHP	100	1020	100	1020
Engine Condition Losses	.15	153	--	--
Reduction Gear Losses	.09	78	.09	92
V-S Linkage Losses	.15	118	.15	139
Blade Condition Losses	.10	67	--	--
Propeller Efficiency Losses	.66	398	.61	481
Hull Efficiency Losses	.10	16	.10	31
Relative Rotative Efficiency Losses	.02	4	.02	6
P.C./EHP	.18	185	.26	271

The net EHP of 185 HP before overhaul does not agree with the 130 HP postulated as a result of the model tests. The most likely cause for this discrepancy is in the displacement assumed for the 7.0 knot trial condition.

There is, of course, some conjecture in the above figures. After considering, weighing, and balancing the various elements, the results appear reasonable. Practical changes that could be made to the hull form to improve the situation are limited and would be of small effect. One such change would be to add a fairing piece to reduce separation in back of the step just aft of the after propulsors. Some improvement in hull resistance would result but the cost would hardly justify it. In summary, short of repowering, the SEACON is probably doing as well now speedwise as can be expected.

END

DTIC

6-86